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Reciprocating Compressor Performance Simulation

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ABSTRACT

An integrated simulation model was developed and implemented for reciprocating compressors. The overall compressor performance was simulated as in a standard calorimeter test system. The analysis focused on compressor instead of system. The simulation model includes heat transfer and flow resistance in compressor components, effect of re-expansion volume, leakage through piston and cylinder clearance, bearing power loss, piston friction loss, oil slosh loss, and valve dynamics. The capacity and efficiency of the compressor were calculated.

Some empirical factors were used in the model, so that the prediction can be more accurate with few test data. A modified blow-by calculation model was proposed. A program module was developed and included in the program for bearing analysis. The performances of R22 and R410A in the compressors with similar mechanical design are simulated and compared to test data. A fairly good agreement was achieved. Some effects of design parameters on compressor performance were investigated. And some data useful to the component design were obtained.

INTRODUCTION

The requirements for the compressor performance become higher and higher due to energy and environmental concerns. And the compressor performance approaches to its practical limit. The further performance improvement and economical component design need better understanding of the compression process, heat transfer, flow resistance, temperature pattern in the compressor components. Therefore, the compressor component and system simulation becomes a useful tool for engineers and researchers in this industry. Several technical papers on these topics have been published in recent years. Most of the simulation models and software emphasized on the system of air conditioner or refrigeration (Arthur et al, 1996 and 1997). Few of them simulated at the component level of the compressor.

There are several factors effecting the compressor performance. Valve dynamics, re-expansion volume, heat transfer, flow resistance in the gas passage, friction loss, and the blow-by through the clearance between piston and cylinder wall are the most significant factors.

Discharge and suction valves are the most critical components for a reliable and high efficient compressor. The discharge and suction valves work under most severe flow and mechanic conditions. The port sizes are restricted by the cylinder bore size and geometry. The delay of valve opening and closing causes back-flow and directly affects the compressor capacity and efficiency.

The re-expansion volume is dependent on the head clearance and the suction and discharge valve geometry. The re-expansion volume has significant effect on the volumetric efficiency and capacity.

The direct effect of the flow resistance in the suction and discharge gas passages is the increase of discharge and suction pressure ratio inside the cylinder for given condensing and evaporating pressures. The flow resistance has some effects on the suction and discharge gas temperature variation, which is insignificant comparing to the direct heat transfer.

There are debates and contradictory opinions regarding to the significance of heat transfer influence on the compressor performance. There are two reasons for the diverse conclusions. The first reason is the complexity of the phenomenon. Different experiments or different models have different results. The second one is different scope of the evaluation. Some researcher only focused on the regenerative heat transfer between the gas and cylinder wall. Others might evaluate overall heat transfer effect. Prasad (1998) gave a good review of the works on heat transfer in reciprocating compressors. Heat transfer effect on the compressor performance is simulated and the results are discussed later in the paper.

Blow-by is another important factor for the performance. Chatzidakis (1997) used an approximate equation from adiabatic model to estimate the blow-by rate. This model didn't count the flow resistance throw the leak passage. So, it may overestimate the leak rate. A modify model is proposed and implemented in this simulation.

The purpose of this simulation is not to provide performance data replacing the performance tests but to provide information for engineers to identify and evaluate the component influence on the overall performance and to design more reliable parts. Another important purpose is to identify the improvement potential for the performance.

MATHEMATICAL FORMULATION

Refrigerant Thermal Property

There are several sets of models and regressions available for R22 and R410A thermal properties. For simplicity, a new set of correlating equations was generated from the data of NIST Standard Reference Database 23 – Version 6.01. The deviations of the correlating equations from the original data of the database are less than 1% within the compressor operating envelope. For some properties, two even three equations are correlated to cover different range of the parameters. For concise, the equation are not listed in this paper.

Thermodynamics

The thermodynamic cycle of the simulated system is shown in Figure 1. Saturation condensing and evaporating temperatures are used to specify the operating condition. A description of the processes in the components of system starts at the exit of evaporator. The state of the refrigerant at exist of the evaporator is superheated gas (1). In the suction tube, suction manifold, and cylinder head, gas pressure decreases due to the flow friction. Temperature increases due to heat transfer from sump refrigerant-lubricant mixture, and discharge gas (1-2). Flowing through the suction valve is an adiabatic process (constant enthalpy) (2-3). The compression inside the cylinders can be considered as polytropic process (3-4). A different polytropic exponent may be used instead of isentropic exponent based on test results to take account of the isentropic efficiency of the compression. The discharge gas flowing through discharge valve is an adiabatic process with pressure drop (4-5). Pressure and temperature decreases as the discharge gas flows through cylinder head, discharge muffler, shock-loop, and discharge line as a result of flow friction and heat transfer (5-6). The superheated

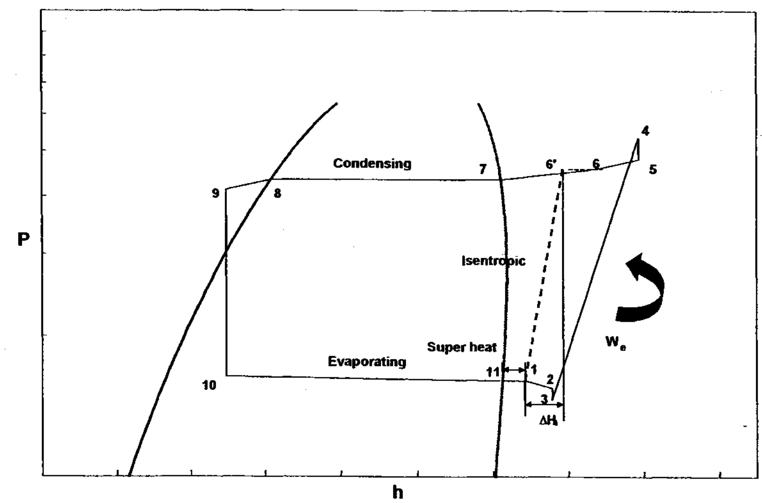


Figure 1 Schematic p-h diagram

discharge gas is cooled down to Saturation State, condensed to liquid, and cooled down to subcooled liquid inside condenser (6-7-8-9). Pressure may decrease through condenser. For the calorimeter system, the shell tube heat exchanger is used. The pressure drop of condensation in shell side can be neglected. The process in expansion device can be considered as adiabatic process with pressure drop from condensation pressure to evaporating pressure (9-10). The process in evaporator includes evaporating and superheating (10-11-1). The pressure drop in the evaporator is evaluated in simulation model. The calculation of pressure drop and temperature increasing or decreasing from heat transfer has been discussed above. Following is the thermodynamic relationship in the cycle.

$T_8=T_c$ $\quad P_8=P_c=P_{sat}(T_c)$ $\quad T_9=T_c-\Delta T_{sc}$ $\quad P_9=P_c$ $\quad P_{10}=P_e=P_{sat}(T_e)$

$h_{10}=h_9$ $\quad P_{11}=P_{10}-\Delta P_{e,tp}$ $\quad T_{11}=T_{sat}(P_{11})$ $\quad P_1=P_{11}-\Delta P_{e,sp}$ $\quad T_1=Te+\Delta T_{sh}$

$P_2=P_1-\Delta P_{suc}$ $\quad T_2=T_1-\Delta T_{suc}$ $\quad P_3=P_2-\Delta P_{sv}$ $\quad h_3=h_2$ $\quad \dfrac{P_3}{\rho_3^\gamma}=\dfrac{P_4}{\rho_4^\gamma}$

$P_4=P_c+\Delta P_{dis}+\Delta P_{dv}$ $\quad P_5=P_c+\Delta P_{dis}$ $\quad h_5=h_4$ $\quad P_6=P_c$ $\quad T_6=T_5-\Delta T_{dis}$

$P_7=P_c$ $\quad T_7=T_c$

The polytropic exponent γ can be measured from experiments. Usually, the isentropic exponent is used.

Heat Transfer

The heat transfer in compressor can be counted as four parts. The first part is suction gas heating by the oil/refrigerant mixture in compressor sump. Second one is discharge gas cooling by oil/refrigerant mixture in compressor sump through muffler, shock loop. Third one is heat transfer from discharge gas to suction gas through cylinder head wall and valve plate for most of reciprocating compressors. Fourth one is regenerating heat transfer from discharge gas to cylinder/piston wall then from cylinder/piston wall to suction gas.

For the last part, the discharge gas heats the cylinder wall and front of the piston. After the cold piston side surface sweeps through the hot cylinder surface and makes the cylinder surface cooler, the cylinder surface contacts with suction gas. Because the piston reciprocating frequency is high, the regenerating heat transfer would be very small relative to other three parts. Therefore, only the first three parts are calculated in this simulation.

In the same way as flow resistance calculation for the flow passages, the first two parts of heat transfer are calculated as steady flow using turbulent pipe flow correlation (Rohsenow, etc, 1997).

The heat transfer inside of head cylinder is unsteady state process due to the pulsate flow. There is no existing experimental correlation to use. Steady state turbulent pipe flow correlation is used for suction and discharge gas flow and then the overall heat transfer coefficient is calculated with the consideration of the conduction resistance.

Flow Resistance

The suction and discharge gas flow passages are simplified into straight tube, bend, and sudden expansion and contraction components. For each component, the common correlating equations (White, 1994) are used to calculate the pressure drop.

Pressure drop in evaporator is calculated the following correlative equation regressed from R410A evaporating data in the literature (Grant, 1999).

$$\frac{\Delta P_e}{L_e}=0.03787F_m^{\ 2}-5.565F_m+695.56$$

For R22, $$\frac{\Delta P_e}{L_e} = 0.02515F_m^{\ 2} + 7.6185F_m - 369.8$$

Where, F_m is mass flux in the tube cross section. And L_e is tube length of the evaporator.

Valve Dynamics

As the first step to initialize the cylinder head pressure (P_5), a modified orifice model is used to estimate the flow resistance of the discharge and suction valves. The model can be stated as:

$$\Delta P = \frac{m^2(1-\beta^4)}{2\rho C_d^2 A^2} \qquad C_d = 0.5959 + 0.0312\beta^{2.1} - 0.184\beta^8 + C_f\beta^{2.5}\,Re^{-3/4}$$

Where, $\beta = d_p/d_v$, and C_f is dependent on the valve design. It can be determined by experiments. It takes the value of 91.71 for the plain orifice. A is the port area.

Using cylinder head pressure calculated from the orifice model, the vibrations of the discharge valve and suction valve are simulated. The governing equation of the valve dynamics can be expressed as follows for the disk type discharge valve.

$$m_v\ddot{x} + c\dot{x} - F_{gas} + kx + F_0 = 0 \qquad F_{gas} = (P_4 - P_5)A_{ef}$$

From the mass balance, and compression process:

$$P_{4,0}\left(\frac{V_0}{m_{g,0}}\right)^{\gamma} = P_4\left(\frac{V}{m_{g,0} - \int_0^t \dot{m}_g dt}\right)^{\gamma}$$

The spring constant k takes very large number when the displacement reaches the seat and lift upper limit.

A_{ef} is called effective force area, which is a function of the valve lift. The equation derived in the reference (Werner Soedel, 1984) is used for discharge valve. The cylinder head pressure (P_5) is assumed to be constant. The mass (m_v) is the sum of valve mass and one third of the spring mass. m_g is the refrigerant mass inside the compression cylinder.

The same model is used for the ring-type suction valve as well. For the ring valve used, the mass m_v takes two third of the valve mass. The spring constant and effective force area as functions of the valve lift can be calculated by finite element analysis using the Pro-MECHANICA.

The damping factor c takes the average value of ~0.014 kg/s for discharge valve and ~0.036 kg/s for suction valve estimated from a vibration testing outside of compressor with consideration of refrigerant gas viscosity (Boswirth, 1990).

Blow-by Rate

The blow-by flow through the clearance between the piston and cylinder wall can be estimated by the following equation from adiabatic duct flow model (Kouremennos, 1980) with friction correction.

$$m_L = K\pi d_p \delta\sqrt{P_4\rho_4}\,\frac{\sqrt{\frac{2Cp}{R}\left(\left(\frac{P_4-\Delta P_l}{P_4}\right)^{2/\gamma}-\left(\frac{P_4-\Delta P_l}{P_4}\right)^{(\gamma+1)/\gamma}\right)}}{\sqrt{1-\left(\frac{P_4-\Delta P_l}{P_4}\right)^{2/\gamma}}},\quad \text{if } P_4\text{-}\Delta P_l>P_5$$

If P_4-ΔP_l<P_5, replace P_4-ΔP_l by P_5. $\Delta P_l = \frac{8\mu m_L L}{\pi\rho d_p \delta^3}$, which is the flow resistance along the leaking passage. K is time fraction of the discharge process. ρ, C_p, R, μ are the gas density, specific heat, gas constant, and viscosity respectively. If the piston rings or flip seals are used, experimental measurement may be needed to determine the blow-by rate.

Power Consumption.

The power consumption can be estimated using the following equation.

$$W = \frac{(m+m_L)(h_4-h_3)+\Delta W_m}{\eta_e} \qquad \Delta W_m = \Delta W_{bf} + \Delta W_{pc} + \Delta W_{om}$$

ΔW_m consists of three parts: bearing friction (ΔW_{bf}), piston/cylinder wall friction (ΔW_{pc}), and power of oil-mix (ΔW_{om}). The bearing friction power consumption (ΔW_{bf}) is calculated through bearing tribology analysis, which will not be discussed in this paper, but the analysis program is part of the modeling program.

The linear velocity profile is assumed for the fluid film between piston and cylinder wall. Piston/cylinder wall friction energy loss is calculated based on this assumption.

$$\Delta W_{pc} = 2\,n\,f\int_0^{\pi}\mu\frac{u_p(\theta)}{\delta}\pi d_p L_P L(\theta)d\theta \qquad L(\theta) = S\sin^2(\theta/2)+\frac{S^2}{8L_{con}}\sin^2\theta$$

$$u_p(\theta) = \pi f S(\sin\theta + \frac{S}{4L_{con}}\sin 2\theta)$$

Where, n, f, S, and q are number of cylinders, motor frequency, stroke, and crank angle respectively. L_p and L_{con} are the piston and connecting rod lengths. d_p is the piston diameter.

The drag force of the throw blocks and offset shaft revolving inside of the oil/refrigerant mixture is calculated based on model of immersed body in fluid stream. Then the corresponding power is estimated based on the revolving speed and throw.

$$F_{df} = \frac{1}{2}C\rho_{oil}U^2A_f \qquad U = \pi f S$$

For turbulent flow, Re>10^4, C is approximate 0.64. A_f is the frontal projected area. U is average sweeping velocity.

$$\Delta W_{om} = nF_{df}U$$

Performance Parameters

The following parameters are calculated in the program to evaluate the compressor performances.

Thermal COP is defined as $COP_t = \frac{m(h_1 - h_{10})}{(m + m_L)(h_4 - h_3)}$

Over all COP includes the mechanical power loss and motor power loss, which can expressed as

$$COP = \frac{m(h_1 - h_{10})\eta e}{(m + m_L)(h_4 - h_3) + \Delta W_m}$$

RESULTS AND DISCUSSION

The program is developed based on a 10-22KW air-conditioning compressor family. The analysis results presented in this paper are for 17 KW R22 and R410A compressors. These two models have the same mechanical configuration but different strokes. The main characteristic parameters of the compressors are listed in Table 1.

Table 1. Compressor characteristic parameters

Parameter	Bore, cm	Stroke, cm	Motor Eff.	A_{dp}, cm^2	A_{sp}, cm^2	V_{ex}, cm3	Lubricant
R22, 17 KW	5.08	2.352	0.901	3.375	5.607	2.622	Mineral
R410A, 17 KW	5.08	1.585	0.927	3.375	5.607	2.622	POE

Where, A_{dp}, and A_{sp} are the discharge and suction port area. V_{ex} is the re-expansion volume.

The simulation results for both compressor models at several conditions are listed in Table 2 and compared with test data where it is available. The sub-cooling and superheat are 8.3°C and 11.1°C respectively for all the simulation and test conditions. From the results, it can be seen that the simulation program predicts the overall performance fairly well.

Table 2 The simulation results.

Condition (°C) (Evap/Cond)		**R22**			**R410A**			
		7.2/54.4 (ARI)	**7.2/43.3**	**7.2/37.8**	**7.2/54.4 (ARI)**	**7.2/43.3**	**7.2/37.8**	**-1.1/43.3**
Capacity, KW	**Pred.**	16.94	19.45	20.73	16.28	19.87	21.70	14.09
	Test	16.91	19.96	21.49	16.28	20.36	22.64	14.14
COP	**Pred.**	3.300	4.541	5.366	3.067	4.586	5.635	3.350
	Test	3.346	4.515	5.295	3.051	4.430	5.429	3.351
Blow-by, kg/s		0.00280	0.00223	0.00197	0.00416	0.00352	0.00314	0.00308
Suct. Side ΔP, kPa		22.70	23.52	23.95	16.07	17.11	17.69	12.57
Valve ΔP, kPa		14.46	14.56	14.62	10.62	10.86	11.02	8.26
Disch. Side, ΔP, kPa		134.16	157.74	172.16	89.63	109.43	122.2	70.46
Valve ΔP, kPa		72.76	74.03	74.43	49.54	51.12	51.92	38.35
Muffler ΔP, kPa		42.66	57.88	67.31	27.94	40.54	48.81	22.30
Mechanical Loss, W		147.36	154.94	159.0	145.77	146.00	148.95	151.81
Electrical Loss, W		508.45	424.17	382.6	387.6	316.30	281.23	307.05

The effects of various parameters on efficiency and capacity are calculated and discussed below.

Re-expansion volume effect

By increasing the re-expansion volume 1% of the displacement, the capacities of the R22 and R410A compressors decrease 3.6% and 4% respectively. The corresponding efficiencies decrease 2% and 2.4% for R22 and R410A compressors.

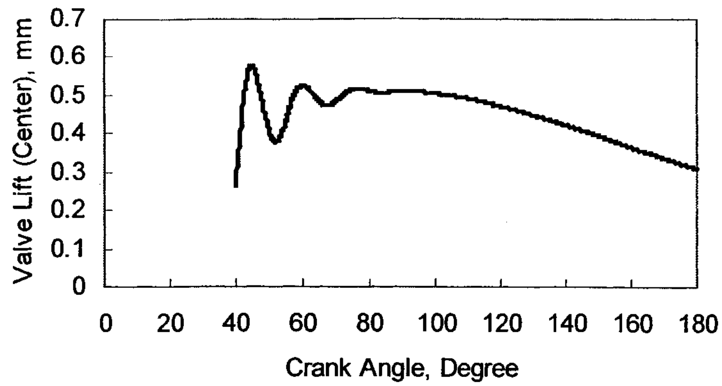
Valve vibration and its effects

For the ARI condition (7.2°C/54.4°C), the discharge valve and suction valve vibrations are displayed on Figures 2. The P-V diagrams are shown in Figure 3. The discharge valve vibration curves for three different spring constants are shown in Figure 2 (b). The effect of discharge valve spring constant on the efficiency is insignificant.

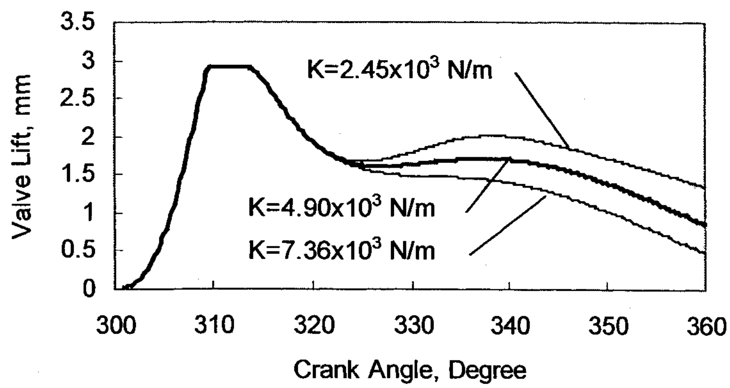
Flow Resistance and Heat Transfer

Examining the flow resistance and heat transfer data in Table 2, it seems that the main flow resistance in the gas flow passages comes from discharge muffler, and valves. The suction valve flow resistance is 63% of the suction side total resistance. The resistances of discharge muffler and discharge valve are 32% and 54% of the total resistance of the discharge side, for R22 compressor at ARI condition.

The internal suction gas heating reduces the capacity 4.26% for 10°C temperature increase for the R22 compressor at ARI condition. And the efficiency decreases 4.32%. The capacity and efficiency for R410A compressor decreases 5.34% and 5.62% respectively under the same condition. The suction gas heating increases the specific volume of the gas, and reduces the mass flow rate consequently for the same volume displacement. The



a) Suction valve vibration



b) Discharge valve vibration

Figure 2 Valve Vibration of R22 compressor at ARI condition

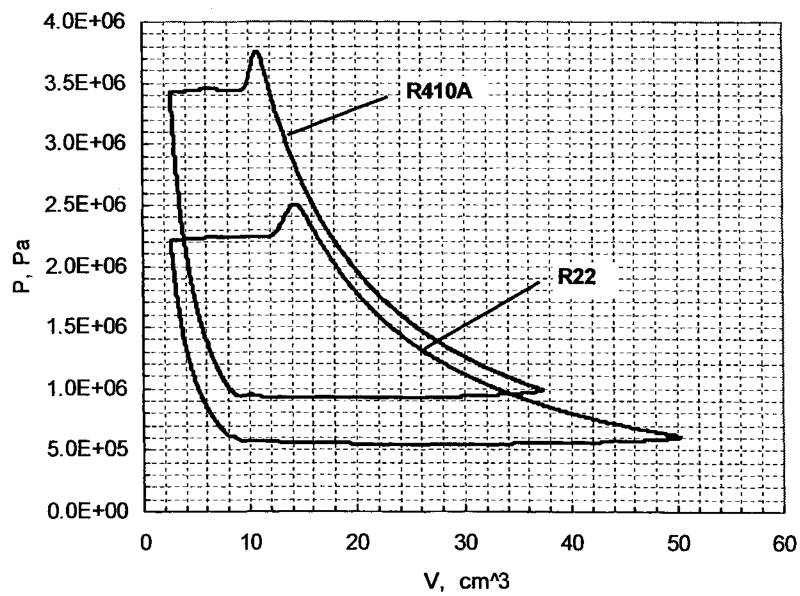


Figure 3 P-V diagrams for R22 and R410A at ARI condition

compression pressure remains approximately the same, so does the power consumption. Therefore, the efficiency would decrease about same percentage as the capacity.

Blow-by

The proposed blow-by rate model successfully predicted the leak rate between the piston and cylinder wall clearance. As seen in Table 2. The leak rates are about 2.6% and 4% for the R22 and R410A compressors at ARI condition. The leak rate decreases as the condensing temperature decreases under the same evaporating temperature for both refrigerants.

CONCLUDING REMARKS

A comprehensive performance simulation program is developed for reciprocating compressor for R22 and R410A refrigerants. The thermodynamic, fluid dynamics, heat transfer, bearing analysis, valve vibration are included in the program. A modified model for leak rate through the clearance between the piston and cylinder was proposed. A discharge and suction valve dynamic model under the operating condition was developed and utilized. The program was verified to be successive on predicting the performance and its variation.

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